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Request for grant of a patent

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1.	Your reference	FP01H03/P-GB HK/JR/IP
2.	Patent application number *(The Patent Office will fill this part in)*	0320022.7 — 27 AUG 2003
3.	Full name, address and postcode of the or of each applicant *(underline all surnames)*	Freepower Ltd. P.O. Box 1642 Salisbury SP2 0SE United Kingdom
	Patents ADP number *(if you know it)*	8700772600l AIL MB 23.3.04
	If the applicant is a corporate body, give the country/state of its incorporation	ENGLAND
4.	Title of the invention	Turbine
5.	Name of your agent *(if you have one)*	Hammonds
	"Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)*	7 DEVONSHIRE SQUARE CUTLERS GARDENS LONDON, EC2M 4YH ~~Rutland House~~ ~~148 Edmund Street~~ ~~Birmingham B3 2JR~~ United Kingdom
	Patents ADP number *(if you know it)*	FSI MB 2.3.04 8621828001 ~~8622~~68000l

6. Priority: Complete this section if you are declaring priority from one or more earlier patent applications, filed in the last 12 months.	Country	Priority application number *(if you know it)*	Date of filing *(day / month / year)*

7. Divisionals, etc: Complete this section only if this application is a divisional application or resulted from an entitlement dispute *(see note f)*	Number of earlier UK application	Date of filing *(day / month / year)*

8. Is a Patents Form 7/77 (Statement of inventorship and of right to grant of a patent) required in support of this request? — YES

Answer YES if:
a) any applicant named in part 3 is not an inventor, or
b) there is an inventor who is not named as an applicant, or
c) any named applicant is a corporate body.
Otherwise answer NO (See note d)

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Patents Form 1/77

9. Accompanying documents: A patent application must include a description of the invention. Not counting duplicates, please enter the number of pages of each item accompanying this form:

Continuation sheets of this form	
Description	12
Claim(s)	2
Abstract	1
Drawing(s)	6 only

10. If you are also filing any of the following, state how many against each item.

Priority documents

Translations of priority documents

Statement of inventorship and right to grant of a patent (Patents Form 7/77)

Request for a preliminary examination and search (Patents Form 9/77)

Request for a substantive examination (Patents Form 10/77)

Any other documents (please specify)

11. I/We request the grant of a patent on the basis of this application.

Signature(s) Hammonds

Date 27 August 2003

12. Name, daytime telephone number and e-mail address, if any, of person to contact in the United Kingdom

Kathleen Harris
0870 839 1374

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Applicant: Freepower Ltd
Attorney's ref: FP01H03/P-GB

DUPLICATE

Turbine

The present invention relates to rotary machine componentry, and more particularly relates to a two-stage radial flow turbine.

Radial inflow turbine systems are well known. Here, the turbine has a set of vanes mounted on a shaft, and a fluid (e.g. a hot, pressurised gas) is incident radially on the vanes of the turbine, causing rotation of the shaft, and afterwards exits in another direction, such as parallel to the shaft axis. Thermal energy in the fluid is transferred into rotational kinetic energy in the shaft.

US-B-4,000,609 discloses a small radial flow gas turbine engine having a small high speed shaft assembly including a central shaft member, a radial outflow compressor rotor secured to one end of the central shaft member, and a radial inflow turbine rotor secured to the other end of the central shaft member.

Axial flow turbines are also well established. For example, US-A-2003/0084900 discloses a respiratory assistance apparatus equipped with a two-stage axial turbine for delivering pressurised gas. The apparatus includes a casing having a gas inlet orifice, a motor having a motor shaft driven by the motor, and a first propeller and a second propeller mounted securely to the motor shaft. An internal partition is arranged in the casing, between the two propellers, so as to form a first turbine stage internal to the casing containing the first propeller, and a second turbine stage internal to the casing containing the second propeller, the first stage and second stage being arranged in series.

Thus, while single stage radial flow turbines are known, and two-stage axial flow turbines are known, a problem is that heretofore there has been a lack of a two-stage radial flow turbine design capable of operating at the high-speed and extremely high pressure differentials encountered in some industries. Often, a problem is that it is not possible for a single stage radial turbine to cope with certain pressure drops.

The present invention provides a radial inflow turbine unit, comprising: a housing with an inlet port for receiving fluid at a first pressure; a shaft mounted on a bearing within the housing and having an axis of rotation; a turbine, disposed on the shaft, the turbine comprising a first turbine stage, comprising a first series of vanes mounted on the shaft, said fluid received by the inlet port being radially incident on said first series of vanes and exiting the first turbine stage at a third pressure and in a first predetermined direction, a second turbine stage, comprising a second series of vanes mounted on the shaft, a conduit for conveying the fluid exiting the first turbine stage to the second turbine stage, said fluid received by the second turbine stage being radially incident on said second series of vanes and exiting the second turbine stage at a second pressure and in a second predetermined direction, wherein said fluid imparts rotational energy to said shaft at both said first and second turbine stages.

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Preferably, the first pressure is about 2 to 10 times the second pressure. Preferably, the third pressure is about 3-4 times the second pressure.

Preferably, the radial dimension of said second turbine stage is greater than the radial dimension of the first turbine stage. Preferably, the radial dimension of second first turbine stage is about 1.25 times the radial dimension of the first turbine stage. Preferably, the axial dimension of said first turbine stage is about 0.3 to 0.375 times the radial dimension of the first turbine stage. Preferably, in the axial dimension of said second turbine stage is about 0.35 to 0.4 times the radial dimension of the second turbine stage.

In a particular embodiment, the turbine unit further includes: a third turbine stage, comprising a third series of vanes mounted on the shaft, a conduit for conveying the fluid exiting the second turbine stage to the third turbine stage, said fluid received by the third turbine stage being radially incident on said third series of vanes and exiting the third turbine stage at a fourth pressure and in a third predetermined direction, wherein said fluid imparts rotational energy to said shaft at said first, second and third turbine stages.

Preferably, the axial dimension of said third turbine stage is about 1/3 times the radial dimension of the third turbine stage.

Preferably, said first, second and/or third predetermined directions is generally axial.

In one embodiment, said fluid is a gas. Preferably, said fluid is HFE-7100 or hexane. The fluid may be one of the alkanes.

The invention further provides an energy recovery system, for extracting energy from a source of waste heat, the system being a closed system with a circulating working fluid, comprising a heat exchanger, an electromechanical conversion unit, a cooling system and a turbine unit according to any of the appended claims, the heat exchanger supplying, in use, the working fluid to said turbine.

Preferably, said fluid permeates the housing, thereby providing lubrication of the bearing.

An advantage of the invention is that it is usable at high rotational speeds (e.g. 25,000 to what 50,000 Rpm). An additional advantage is that the two-stage design entails a pressure drop occurring at each stage, allowing it to cope with higher input pressures (e.g. up to 20 bar absolute).

A further advantage is that a relatively compact design of the turbine is pe^t^rmitted.

The foregoing attributes ensure that the turbine may advantageously be employed in systems (e.g. Rankine cycle systems) where energy conversion occurs from fluids (gases) at very high operating pressures, with improved efficiency.

The present invention will now be described, by way of example, with reference to the accompanying drawings in which:

Figure 1 is shows (a) schematic overview of an energy recovery system in accordance with one aspect of the invention, and (b) intermediate electronics modifying the output of the alternator;
Figure 2 is a schematic illustration of the derivation of one source of waste in one aspect of the invention;
Figure 3 illustrates in more detail the turbine unit and alternator of Fig. 1;
Figure 4 is an enlarged view of the turbine bearing in Fig. 3;
Figure 5 shows in more detail the bearing member employed in the bearing in Fig. 4, indicating fluid flows; and
Figure 6 illustrates an alternative (magnetic) coupling of the turbine unit and alternator of Fig. 1, in another aspect of the invention.

Turning to the drawings, Fig. 1(a) is a schematic overview of an energy recovery system in accordance with one aspect of the invention. A main heat exchanger 102 has at least one source fluid inlet 104 through which it receives a heated source fluid incorporating the thermal energy that is sought to be recovered by the system. The temperature of the source fluid upon entering the main heat exchanger 102 is designated t1.

The main heat exchanger 102 may be driven by any source of heat, and examples of the source fluid include hot air, steam, hot oil, exhaust gases from engines, manufacturing process waste hot fluid, etc. Alternatively, the heat source may be solar thermal energy that heats a suitable fluid (e.g. heat transfer oil) that forms the source fluid for the main heat exchanger 102.

Referring briefly to Fig. 2, this is a schematic illustration of the derivation of one source of waste in one aspect of the invention: an important example of wasted energy is the ubiquitous internal combustion engine, be it petrol, diesel or gas fuelled, reciprocating or turbine. The best simple cycle fossil fuelled engine (other than very large power stations or marine engines) will be between 35-40% efficient: this means that 60-65% of the energy from the fuel used to drive the engine is lost as waste heat.

Returning to Fig. 1(a), the source fluid exits the main heat exchanger 102, at a reduced temperature t2, via at least one source fluid outlet 106.

The main heat exchanger 102, which is suitably of the cross counter flow type, also has a working fluid inlet 108 and working fluid outlet 110, through which it receives (as a liquid at temperature t3) and despatches (at temperature t4), respectively, the working fluid of the system. The working fluid, which is heated and vapourised within the main heat exchanger 102, is carefully chosen so that its thermodynamic and chemical properties are suitable to the system design and the operational temperatures and pressures. In one embodiment, the working fluid is HFE-7100.

After exit from the working fluid outlet 110 of the main heat exchanger 102, the gaseous working fluid flows in the direction of arrows A to the turbine inlet 112 of turbine unit 114. The working fluid arrives at the turbine unit 114 at pressure p1, loses heat and pressure in driving the turbine (not shown) mounted on turbine shaft 116 within the turbine unit 114, and exits the turbine unit 114 via turbine outlets 118 at pressure p2, which is substantially lower than p1. In one embodiment, the pressure p1 is 11.5 bar absolute and the pressure p2 is 1.0 bar absolute.

In one embodiment, the turbine shaft 116 is mounted on a bearing (not shown) and is mechanically coupled to an alternator 120, e.g. the turbine and alternator armature (not shown) are mounted on a common shaft 116. In this way, high-speed rotation of the turbine shaft 116 causes electrical energy to be generated in the alternator 120, the consequent voltage appearing at the alternator output 122. The coupling of the turbine shaft 116 to the alternator 120 is described further hereinbelow with reference to Figs 3 to 5.

After exit from the turbine outlets 118, the working fluid travels in the direction of arrows B to inlet 124 of a second heat exchanger 126, which acts as a preheater of the working fluid using the turbine exhaust. The working fluid is therefore input to the second heat exchanger 126 at temperature t5 and exits via outlet 128 at a lower temperature t6. At the same time, the second heat exchanger receives another flow of working fluid (in the direction of arrows C), below its boiling point and in liquid form, via inlet 130 at temperature t7. In the second heat exchanger 126, thermal energy is transferred to the flow of working fluid arriving at inlet 130, the working fluid exits via outlet 132 at temperature t3, and flows (in the direction of arrows D) to the inlet 108 of the main heat exchanger 102.

The system also includes a condensing unit (or water cooler) 134, in which cold water arrives via inlet 136 and exits via outlet 138. In operation, working fluid from the second heat exchanger 126, flowing in the direction of arrow E, arrives in the condensing unit 134 via inlet 140, is cooled and condensed into a liquid in the condensing unit 134, and then departs via outlet 142. This liquid working fluid (at temperature t7), is forced by pump 144 via valve 146 in the direction of arrows C and forms the second supply of working fluid arriving at second heat exchanger 126, to begin the cycle all over again. In one embodiment, a separate fluid line 160 delivers liquid working fluid to the bearing coupling the turbine unit 114 and the alternator 120, for lubrication.

Thus, the system operates on a Rankine cycle and is sealed, so that there is no escape or consumption of the working fluid, which simply cycles through its various phases.

In one embodiment, the system includes a control system 150, to control the power output by the system. Most existing Rankine cycle machines are low speed units with synchronous alternators, running at the same frequency as the grid supply. Turbine speed and power control is generally by valves to bypass the turbine. However, the system according to one aspect of the present invention

employs a high-speed alternator 120, and a power-conditioning unit is preferably used to convert the high frequency alternator output to mains frequency.

More specifically, the control system includes intermediate electronics 151, a power conditioning unit (PCU) 152 and a controller 154. The power output by the alternator 120 at outputs 122 is at a very high frequency, due to the high-speed rotation of the turbine shaft, and is modified by intermediate electronics 151, which is described in more detail in Fig. 1(b).

Referring to Fig. 1(b), the outputs 122 of the alternator 120 are connected to the inputs 160 (three of them, for a 3-phase alternator) of intermediate electronics, generally designated 151. The first stage of intermediate electronics 151 is an optional transformer stage 162, for boosting the voltage on each of the lines: this ensures, when needed, that there is sufficient dc voltage eventually appearing at the PCU 152 that a complete 240 V sine wave (as per UK mains supply) can be generated at the output of the PCU 152. In certain embodiments, however, the voltage level output by the alternator 120 is high enough such that the transformer stage 162 can be omitted.

Next, the voltages output by the transformer stage 162 at 164 pass to a rectification stage 166, comprising a set of six rectification diodes 168, as is well known in the art. Thus, a rectified, near dc voltage is supplied at outputs 170 of the rectification stage 166, and this, in normal operating conditions appears at the outputs 172 of the intermediate electronics 151.

In the event of a sudden loss of grid connection all alternator load will be lost. This could cause a significant overspeed of the alternator 120, and so as well as a dump valve (not shown) to bypass the turbine, the intermediate electronics 151 includes a safety stage 174 that includes a dump resistor 158 to supply a load to the alternator 120 in the event of loss of grid connection, to prevent overspeed.

A transistor 176 is in series with the dump resistor 158 across the outputs 172, with the base b of the transistor 176 being driven by an overspeed detection unit (not shown). The latter supplies a PWM signal to the transistor 176, the duty cycle of which is proportional to the extent of overspeed, so that the higher the overspeed the greater the load applied by the dump resistor 158.

As can be seen in Fig. 1(b), the power supplied at outputs 172 (referred to herein as dc bus) is at voltage V and current I, and this is supplied to the PCU 152. The PCU 152, which is known in the art, is adapted to convert power from dc to ac at the mains frequency (50 Hz in UK) and voltage (240 V in UK). The PCU in turn is able to vary the dc bus voltage so as to adjust the power output of the system.

Varying the dc bus voltage (V in Fig. 1(b)) in the power conditioning unit 152 controls the speed of the turbine shaft 116. Reducing the bus voltage increases the load on the alternator 120, causing more current to be drawn from the alternator. Conversely, increasing the bus voltage causes the alternator current to drop. By calculating the power (e.g. using P=VI, or using a power measuring device) before and after the bus voltage change, it can be determined whether the power was increased or

decreased by the bus voltage change. This allows the point of maximum power output from the alternator 120 to be found and then continually 'tracked' by altering the bus voltage.

In one embodiment, the voltage supplied by the alternator at no load is 290Vac (all voltages are measured line-to-line) on each of the three phases at 45,000rpm, the maximum rated speed of the alternator 120. The lowest speed at which power can be generated is 28,000rpm, at which point the voltage is 180 Vac at no load. Increasing the load will also reduce the alternator voltage: for example at 45,000 rpm the voltage will be 210 Vac at 6.3 kW.

The control of power output by varying the bus voltage may be implemented by suitable analog or digital electronics, microcontroller, or the like. It may also be controlled manually using a personal computer (PC) as the controller 154. Preferably, however, the power output is controlled automatically using a suitably programmed PC or other computing machinery as the controller 154. In either case, the PC communicates with the PCU 152 by means of a RS232 serial communications device, although a RS422 or RS485 adapter could also be used, as is known in the art. The PC may thus, at any time, have a reading of V and I, thereby enabling the instantaneous power to be known.

In the case of automatic PC control, the method of control may be by means of suitable software implementing the following.

```
While system is ON do
Increase bus voltage by one voltage step
Measure new power (=VI)
if new power less than or equal to old power then decrease voltage by one voltage step
        do
        decrease voltage by one voltage step
        measure new power
        while new power more than old power
else    do
        increase voltage by one voltage step
        measure new power
        while new power more than old power.
```

It will be appreciated by persons skilled in the art that the size of the voltage step is determined by operating conditions and is a suitably determined small fraction (e.g. 1-2.5%) of the mean bus voltage. In on embodiment, the voltage step change is made about every second.

One other optional feature incorporated in the system is a working fluid purification system, generally designated 170 in Fig. 1. As mentioned hereinabove, if there are non-condensable gases present during the running of the system, overall performance can be substantially reduced, i.e. the pressure ratio of the turbine is lower than it should be. For example, in the turbine mentioned in the examples

herein, the input pressure p1 is projected to be 20 bar; and if the output pressure p2 is 2 bar rather than the intended 1 bar, the pressure ratio is 10 rather than 20, giving significantly reduced performance.

A difficulty is that when filling the system initially, the working fluid is a liquid and the rest of the system must be filled with a gas, for example nitrogen. When performing this step the pressure can be reduced to below atmospheric pressure to reduce the mass of nitrogen in the system. However, the pressure cannot be reduced too much, or cavitation will occur in the pump. Therefore, the optimum way to remove the unwanted gas from the system is during the running of the system.

The working fluid purification system 170 includes a conduit 172 connected at one end to a point Q on the second heat exchanger (preheater) 126 and at the other end to control valve 174 which may be at the base entry/exit port 176 of an expansion tank 176, which, in one example, may be the type of expansion tank used in central heating systems. The expansion tank 176 has a flexible membrane or diaphragm 178 so that it may in its lower part contain a variable volume V of gas and/or liquid.

In the example (6kW system) mentioned hereinafter, the measurements are as follows.

System volume	70 litres
Fluid volume	18 litres
Expansion tank volume	50 litres

As can be seen, when the system is initially filled with fluid, there will be 52 litres of nitrogen. Lowering the pressure of this gas with a vacuum pump reduces the amount of gas that has to be held in the expansion tank 176, meaning that it can be made smaller. This pumping also causes the diaphragm 178 expand downwards into the expansion tank, making the whole of the tank, or nearly all of it, available for receiving gases.

As nitrogen gas has a lower density than that of the working fluid vapour, it tends to accumulate at the highest location within the system. At this point (Q in Fig. 1), the fluid can be taken away to the expansion tank 176, the diaphragm 178 allowing expansion to take place, enlarging volume V; i.e., with the control valve 174 open, the gases are allowed to move slowly into the expansion tank 176. As the nitrogen has a lower density than the working fluid, most of the contents of the expansion tank 176 will be nitrogen, with just a little working fluid.

Once the valve 174 has closed, the expansion tank 176 and its contents cool down naturally, causing the working fluid to condense. The next time the control valve 174 is opened, the (now liquid) working fluid flows back under gravity back into the main circuit of the system (via control valve 174 and conduit 172)), while the non-condensable gases tend to stay in the expansion tank 176 due to their lower density. A cycle of (a) control valve OPEN for a fixed period, followed by (b) control valve CLOSED for a fixed period is used to purify the working fluid, and this cycle may be repeated several times (for example about 3 to 5 times), during the start up of the energy recovery system, to collect as

much nitrogen in the expansion tank 176 as possible. In the aforementioned (6kW) system, the control valve 174 is opened for one minute and then closed for ten minutes. The opening and closing of the control valve 174 may be performed manually, or it may be performed automatically by a suitable controller, in this case controller 154.

The system preferably also includes a pressure sensor coupled to the controller 154, the pressure sensor being positioned to sense the pressure at the exit of the expansion device (turbine); and the purification cycle may be repeated if pressure starts to build up during normal running of the system and it is detected at the pressure sensor that the pressure has exceeded a predetermined safe threshold.

Figure 3 illustrates in more detail the coupling of the turbine unit and alternator of Fig. 1(a). Here, the turbine unit is generally designated 114 and the alternator generally designated 120. The turbine shaft rotates about an axis 302 and is integral with a section 304 that forms part of the rotor 306 of the alternator 120. Generally partial cylinder permanents magnets 308 are disposed on the section 304 of the shaft 116. Retaining the magnets 308 in position on the shaft 116 is a retaining cylinder 309: this retaining cylinder (made of a non-magnetic material such as CFRP) ensures that the magnets 308 are not dislodged during high-speed rotation of the shaft 116. The stator 311, incorporating a plurality of windings (not shown) in which current is generated, is mounted around the rotor 306, as is well known in the art, and is enclosed within housing 310. The section 304 of the shaft 116 is supported at one end of the housing 310 by journal bearing 312, and at the other end by the bearing generally designated 314, which is described in more detail hereinafter.

Figure 4 is an enlarged view of the turbine–bearing coupling in Fig. 3. As can be seen, the turbine unit 114 includes a first turbine stage 402 and a second turbine stage 404. High pressure heated working fluid present (at pressure p1) in the space 406 within the turbine unit housing 408 enters via inlet port 410 of the first turbine stage 402 and flows in the direction of arrow F so as to be incident upon a first series of vanes 412 securely mounted on the shaft 116. The fast flowing working fluid thereby imparts rotational energy to the shaft 116. Upon exiting the first turbine stage 402 (at pressure p3), the working fluid flows in the direction of arrows G.

Next, the working fluid at (intermediate) pressure p3 (which is substantially less than p1, but still relatively high) passes, via conduit 413, to the next turbine stage 404. Here, the working fluid enters via inlet port 414 of the second turbine stage 404 and flows in the direction of arrow H so as to be incident upon a second series of vanes 416 securely mounted on the shaft 116. The fast flowing working fluid thereby imparts further rotational energy to the shaft 116. Upon exiting the second turbine stage 404 (at pressure p2), the working fluid flows in the direction of arrow J. Thus, p1 > p3 > p2.

As can be seen, the axial and radial dimensions of the vanes 416 of the second turbine stage 404 are greater than those of the vanes 412 of the first turbine stage 402. In one embodiment, there are two

turbine stages of equal diameter, and the axial dimension of the first turbine stage is 3/10 of the diameter, and the axial dimension of the second turbine stage is 4/10 the diameter. In another embodiment, there are three turbine stages. The diameters of the first, second and third turbine stages are in the ratio 4 : 5 : 6. The axial dimension of the first turbine stage is 0.375 x the respective diameter. The axial dimension of the second turbine stage is 0.35 x the respective diameter. The axial dimension of the first turbine stage is 0.33 x the respective diameter.

A washer 418 is provided fixedly attached to a shoulder 420 of the turbine stage 404 and has its other surface abutting a bearing member 422, which is described in more detail hereinafter, and in operation, the working fluid permeates the space between the washer 418 and the bearing member 422, so as to provide lubrication.

The bearing member 422 has a generally T-shaped cross-section. It includes a first bearing surface 424 on a raised portion on the top of the T; and in use, this surface is disposed opposite a second bearing surface 426, of substantially the same annular shape and size, on the shaft 116 near the armature section 304. The bearing member 422 has a central cylindrical channel 428, thereby defining a cylindrical third bearing surface 430 on bearing member 422 that is disposed opposite fourth bearing surface 432 on the outside of shaft 116. A fifth bearing surface 434 is provided on the bearing member 422 on the end thereof opposite the first bearing surface 424, and this is disposed opposite a respective surface of the washer 418. In one embodiment, the working fluid permeates all the spaces defined opposite bearing surfaces 424, 430 and 434 of bearing member 422, thereby providing lubrication of the bearing. In one embodiment, the working fluid is provided as a liquid from the pump 144 (see Fig. 1(a)) via a fluid pipe 160, separate from the main flows, communicating with the outer surface of the bearing member 422.

It will be appreciated that the bearing in this form provides a bi-directional thrust bearing: the bearing member 422 has two bearing surfaces 424 and 434, enabling it to receive thrust in two directions.

Figure 5 shows in more detail the bearing member 422 employed in the bearing in Fig. 4, indicating fluid flows. Figure 5(a) is an end view showing the first bearing surface 424. The flange 502, forming the top of the T, is provided with two screw holes 504 enabling the bearing member 422 to be screwed or bolted to the housing 310 of the alternator 120. Six equally spaced radially extending first elongate recesses (slots) 506 are disposed in the first bearing surface 424, extending from radial inner extremity of the first bearing surface 424 towards the outer radial extremity of the firs bearing surface 424, enabling the passage of lubricant fluid. As can be seen in Fig. 5(b), each recess 506 does not quite reach the outer extremity 508 of the first bearing surface 424. In the embodiment of Fig. 5(a), each recess 506 is provided with an axially extending second lubrication channels 510, which extend to a circumferential recess (or groove) described hereinafter.

In other embodiments, there may not be a second lubrication channel 510 in each recess 506: for example, Fig. 5(c) illustrates the case where a second lubrication channel 510 is provided in only two of the recesses 506.

Referring to Fig. 5(d), a circumferentially extending recess (groove) 512 is provided in the outer surface 514 of bearing member 422. It can been seen that first lubrication channels 516 (here, four of them, equally circumferentially spaced) extend between the circumferentially extending recess 512 and the interior of the bearing member 422, allowing passage of lubrication fluid. As best seen in Fig. 5(e), the second lubrication channels 510 extend between the first bearing surface 424 and the circumferential recess 512. The ends of the second lubrication channels 510 are also shown in Fig. 5(f). The latter figure also shows a plurality (here six) of second elongate recesses (slots) 516 disposed in the fifth bearing surface 434. Two of the second elongate recesses 516 have second lubrication channels extending therefrom to the aforementioned circumferential recess 512. Figure 5(g) is a partial cross-section showing the recesses and channels in another way.

Returning to Fig. 5(e), the lubrication fluid enters the bearing member 422 in the direction of arrows K. The fluid flows in the direction of arrows L to the first elongate recesses 506 on the first bearing surface 424, in the direction of arrow M to the second elongate recesses 516 on the fifth bearing surface 434, and in the direction of arrow N (into the paper) to the interior of the bearing member and the third bearing surface 430, thereby lubricating the bearing.

Example 1

The specific values for one example (6kW version) of the system are set out below. All pressures are in bar (absolute). All temperatures are in °C. The working fluid is HFE-7100.

t1	t2	t3	t4	t5	t6	t7
150.0	123.4	111.0	165.0	130.0	65.0	55.0

p1	p2	p3
11.5	1.0	3.4

Example 2

The specific values for a second example (120kW version) of the system are set out below. All pressures are in bar (absolute). All temperatures are in °C. The working fluid is hexane.

t1	t2	t3	t4	t5	t6	t7
225.0	138.8	123.8	210.0	145.9	74.0	64.0

p1	p2	p3
19.5	1.0	-

The results from the system demonstrate a very useful thermodynamic efficiency (usable electricity out to heat in) for the heat recovery and solar thermal industries — 10% for a source fluid input at 110°C to 22% for a source fluid input at 270°C.

Referring to Figure 6, this illustrates an alternative (magnetic) coupling of the turbine unit and alternator of Fig. 1(a), in another aspect of the invention. The view in Fig. 6(a) is an axial cross-section of the coupling, showing a first rotary member 602 formed of turbine shaft 604 and a first magnetic member 606. In turn, the first magnetic member 606 comprises an armature portion 608, made of steel or iron, integral with the shaft, and a plurality of magnet sections 610, to be described further hereinbelow.

The first rotary member 602 is hermetically sealed inside a housing 612 that contains the turbine (not shown) and working fluid, the housing including a cylindrical portion 614 containing the first magnetic member 606. At least the portion 614 is made of a non-magnetic material, such as stainless steel, nimonic alloy or plastic.

A second rotary member 616 comprises a second shaft 618 and a generally cylindrical second magnetic member 620 integral therewith. The second magnetic member in turn comprises an outer supporting member 622 having a plurality of second magnet sections 624 fixedly attached to the interior thereof.

As best shown in Fig. 6(b), the first rotary member 602 may have a composite containment shell 626 around at least the cylindrical part thereof, so as to maintain the first magnet sections 610 in place during high-speed rotation. The containment shell may be made of a composite such as carbon fibre reinforced plastic (CFRP), kevlar, or glass fibre reinforced plastic (GRP).

Figure 6(c) is a transverse cross-section at A-A in Fig. 6(a). This shows the first magnet sections 610 and second magnet sections 624 in more detail: in this case there are four of each. The magnet sections are elongate, with a cross-section similar to the sector of a disc. The magnet sections are permanent magnets formed of a suitable material, such as ferrite material, samarium cobalt or neodymium iron boron. The direction of the North-South direction for the magnet sections is radial, as schematically illustrated in Fig. 6(d).

Turning to Fig. 6(e), this shows an alternative embodiment, in which the first magnetic member 606' and the second magnetic member 620' are substantially disc-shaped. The first magnetic member 606' comprises a first mounting section 628 and first magnet sections 610', and the second magnetic member 620' includes a second mounting section 630 and second magnet sections 624'. As before, a non-magnetic portion 614' of the housing (similar to 614 and made of the aforementioned non-magnetic material) separates the faces of the disc-shaped magnetic members 606' and 620', which are in close proximity.

The arrangement of the poles for the magnet sections one or both of the first and second magnetic members 606', 620' is illustrated schematically in Fig. 6(f). As also illustrated in Fig. 6(g), the polarity of the face of the magnet sections 610' (or 624') alternates as you go tangentially from magnet section to magnet section.

These magnet arrangements permit coupling and transfer of rotational energy and torque from the turbine shaft 604 to the shaft 618 of the alternator, and are adapted to do so at relatively high speeds, e.g. 25,000 to 50,000 rpm.

Claims:

1. A radial inflow turbine unit, comprising:
a housing with an inlet port for receiving fluid at a first pressure;
a shaft mounted on a bearing within the housing and having an axis of rotation;
a turbine, disposed on the shaft, the turbine comprising
a first turbine stage, comprising a first series of vanes mounted on the shaft, said fluid received by the inlet port being radially incident on said first series of vanes and exiting the first turbine stage at a third pressure and in a first predetermined direction,
a second turbine stage, comprising a second series of vanes mounted on the shaft,
a conduit for conveying the fluid exiting the first turbine stage to the second turbine stage,
said fluid received by the second turbine stage being radially incident on said second series of vanes and exiting the second turbine stage at a second pressure and in a second predetermined direction,
wherein said fluid imparts rotational energy to said shaft at both said first and second turbine stages.

2. The turbine unit of claim 1, wherein the first pressure is higher than the third pressure, and the third pressure is higher than the second pressure.

3. The turbine unit of claim 1 or 2, wherein the first pressure is about 2 to 10 times the second pressure.

4. The turbine unit of any of claims 1 to 3, wherein the third pressure is about 3-4 times the second pressure.

5. The turbine unit of any of the preceding claims, wherein the radial dimension of said second turbine stage is greater than the radial dimension of the first turbine stage.

6. The turbine unit of claim 5, wherein the radial dimension of second turbine stage is about 1.25 times the radial dimension of the first turbine stage.

7. The turbine unit of any of the preceding claims, wherein the axial dimension of said first turbine stage is about 0.3 to 0.375 times the radial dimension of the first turbine stage.

8. The turbine unit of any of the preceding claims, wherein the axial dimension of said second turbine stage is about 0.35 to 0.4 times the radial dimension of the second turbine stage.

9. The turbine unit of any of the preceding claims, further including:
a third turbine stage, comprising a third series of vanes mounted on the shaft.

a conduit for conveying the fluid exiting the second turbine stage to the third turbine stage, said fluid received by the third turbine stage being radially incident on said third series of vanes and exiting the third turbine stage at a fourth pressure and in a third predetermined direction,

wherein said fluid imparts rotational energy to said shaft at said first, second and third turbine stages.

10. The turbine unit of claim 9, wherein the axial dimension of said third turbine stage is about 1/3 times the radial dimension of the third turbine stage.

11. The turbine unit of any of the preceding claims, wherein said first, second and/or third predetermined directions is generally axial.

12. The turbine unit of any of the preceding claims, wherein said fluid is a gas.

13. The turbine unit of any of the preceding claims, wherein said fluid is HFE-7100 or hexane or water.

14. The turbine unit of any of claims 1 to 12, wherein said fluid is one of the alkanes.

15. The turbine unit of any of the preceding claims, wherein said fluid permeates the housing, thereby providing lubrication of the bearing.

16. The turbine unit substantially as hereinbefore described with reference to the accompanying drawings.

17. A waste energy recovery system, for extracting energy from a source of waste heat, the system being a closed system with a circulating working fluid, comprising a heat exchanger, an electromechanical conversion unit, a cooling system and a turbine unit according to any of the preceding claims, the heat exchanger supplying, in use, the working fluid to said turbine unit.

Applicant: Freepower Ltd
Attorney's ref: FP01H03/P-GB

Abstract

Turbine

A radial inflow turbine unit, comprising: a housing with an inlet port for receiving gas at a high pressure; a shaft mounted on a bearing within the housing and having an axis of rotation; a turbine, disposed on the shaft, the turbine comprising a first turbine stage, comprising a first series of vanes mounted on the shaft, said gas received by the inlet port being radially incident on said first series of vanes and exiting the first turbine stage at a third (intermediate) pressure, a second turbine stage, comprising a second series of vanes mounted on the shaft, a conduit for conveying the gas exiting the first turbine stage to the second turbine stage, said gas received by the second turbine stage being radially incident on said second series of vanes and exiting the second turbine stage at a second pressure (which may be about an order of magnitude lower than the inlet pressure), wherein said gas imparts rotational energy to said shaft at both said first and second turbine stages. In one embodiment, a third turbine stage is provided downstream of the second turbine stage. The design permits high rotational speeds and high efficiency in transferring energy to an alternator.

(Fig. 4)

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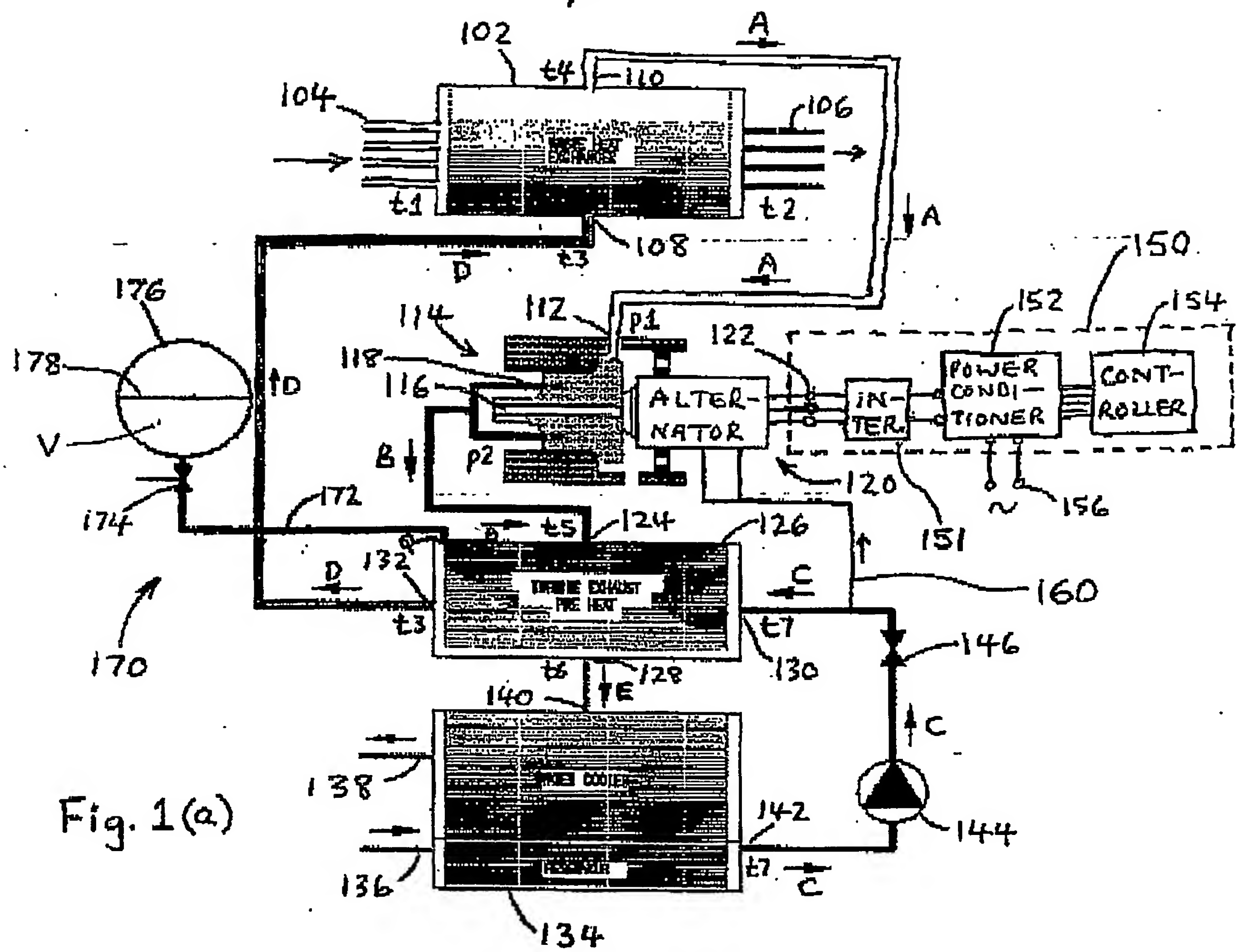

Fig. 1(a)

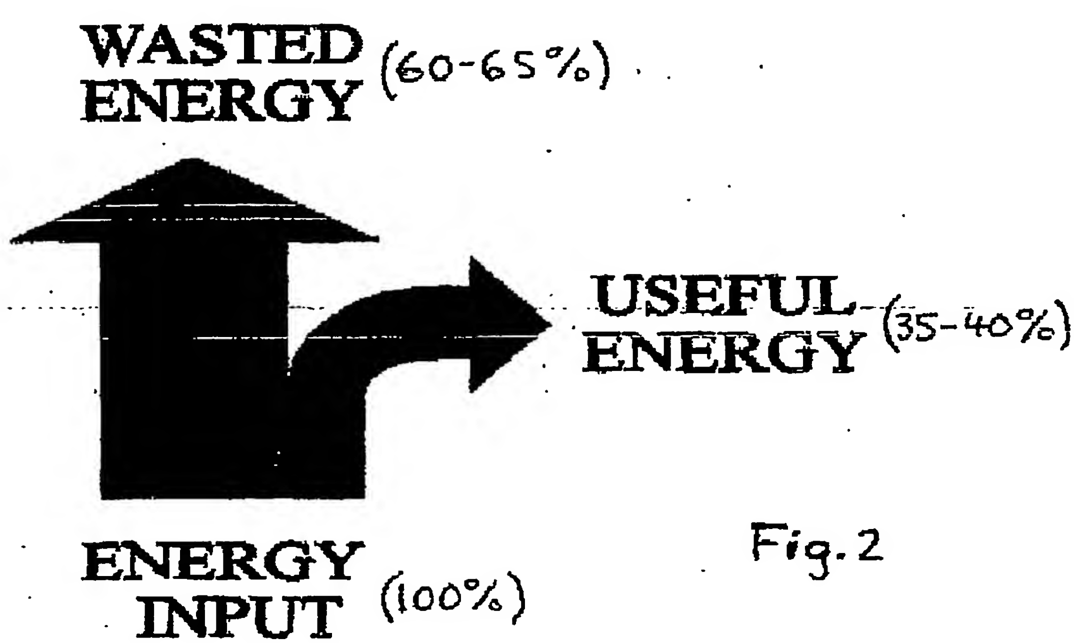

Fig. 2

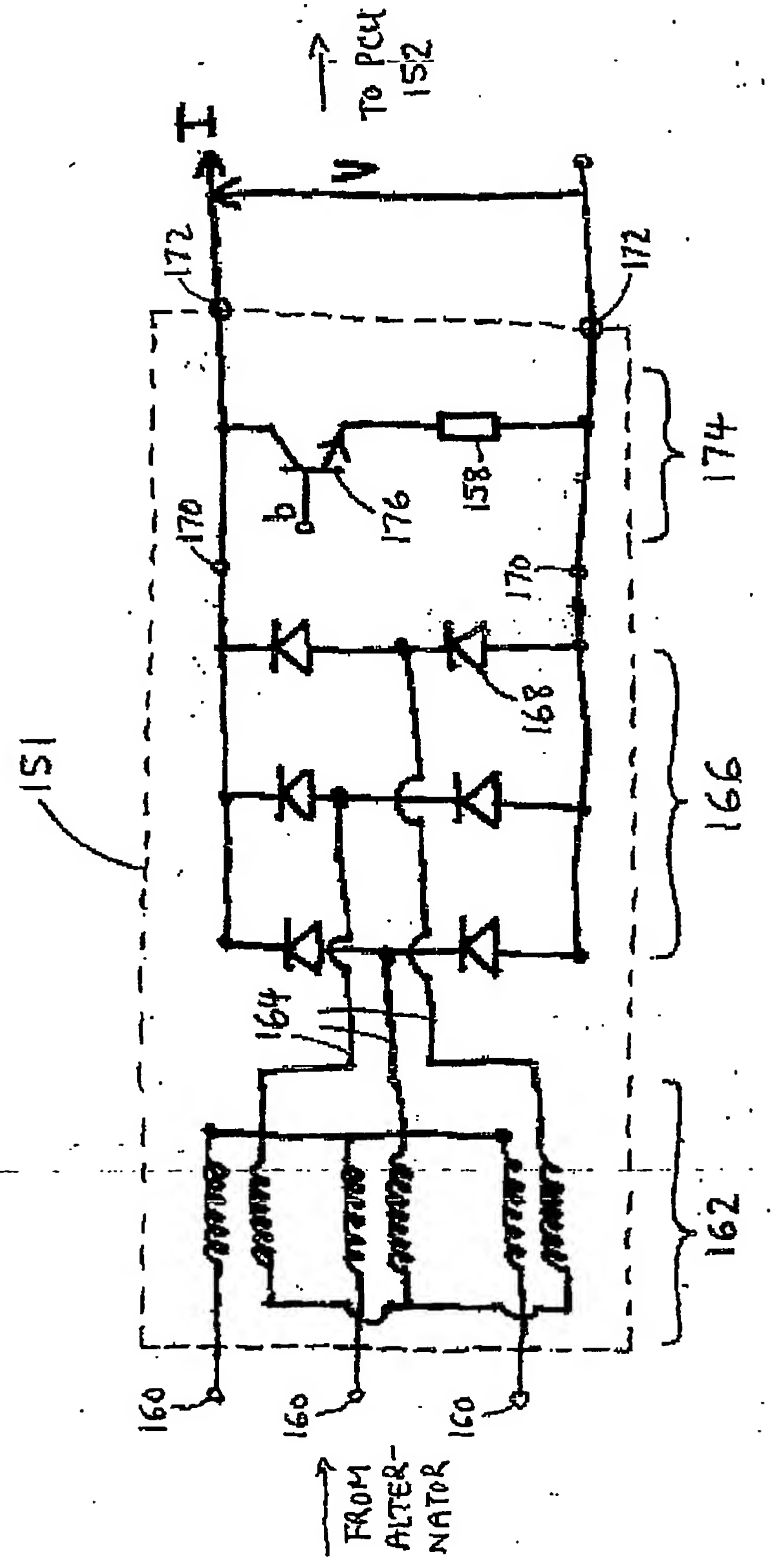
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Fig. 1(b)

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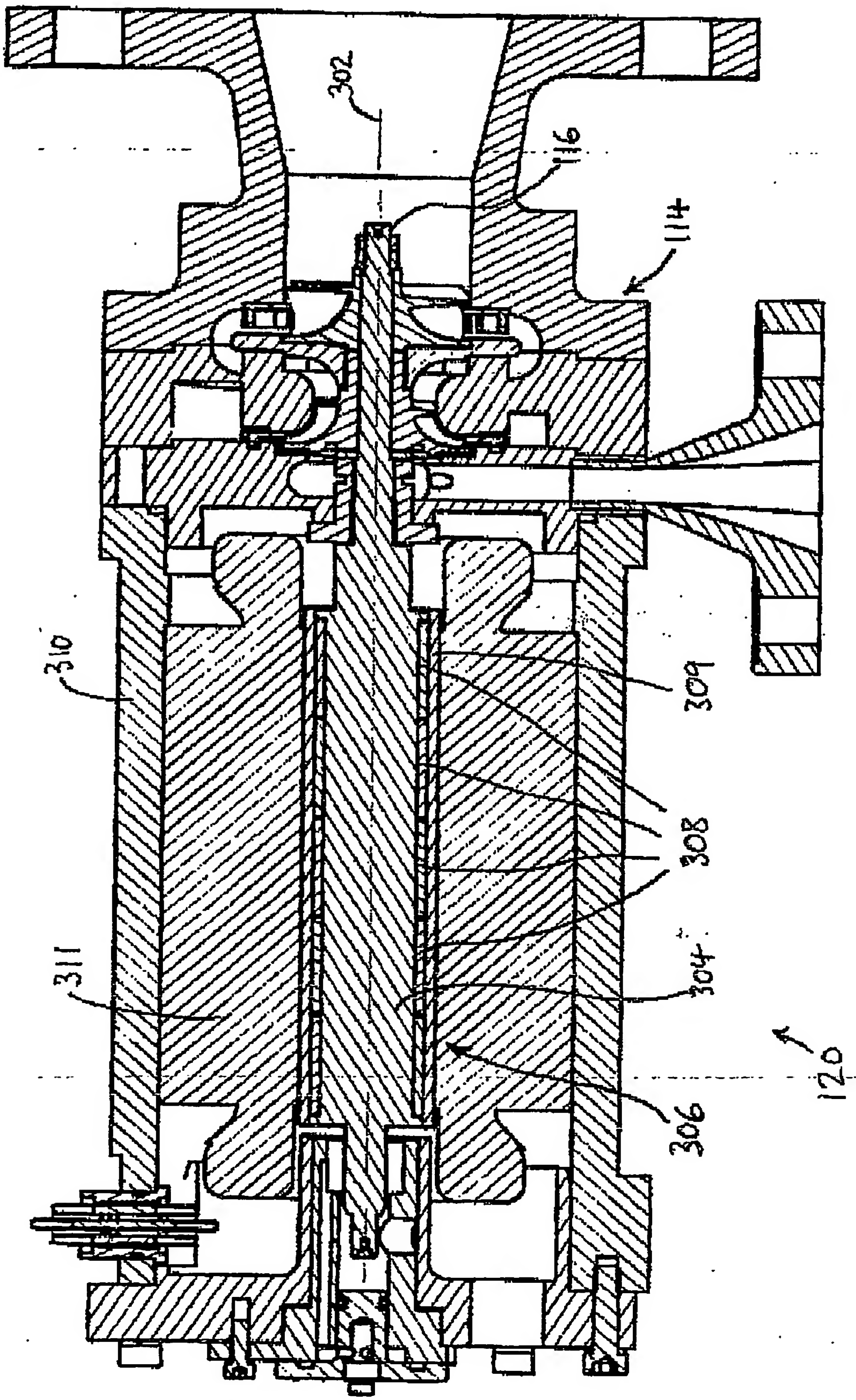

Fig. 3

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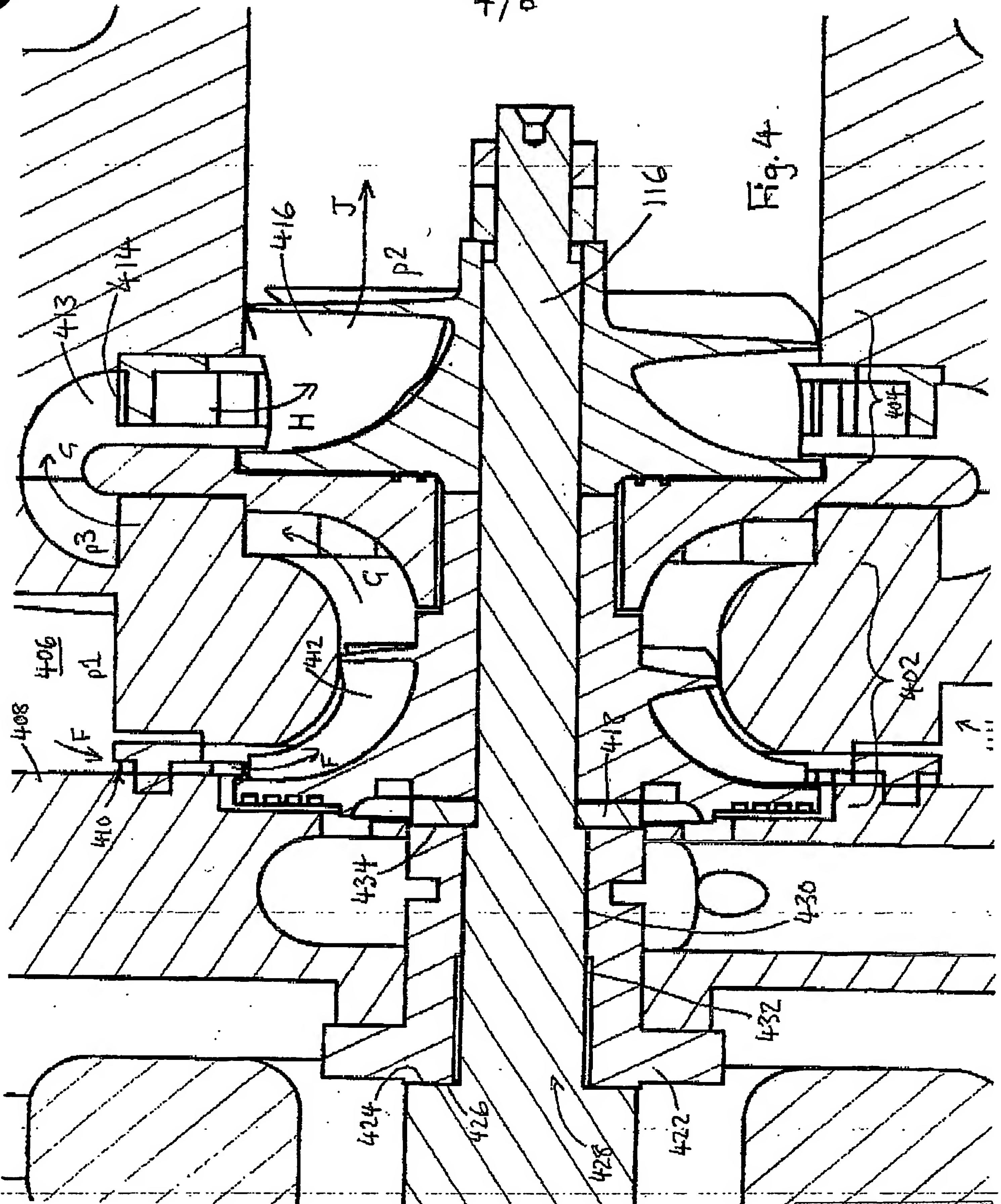
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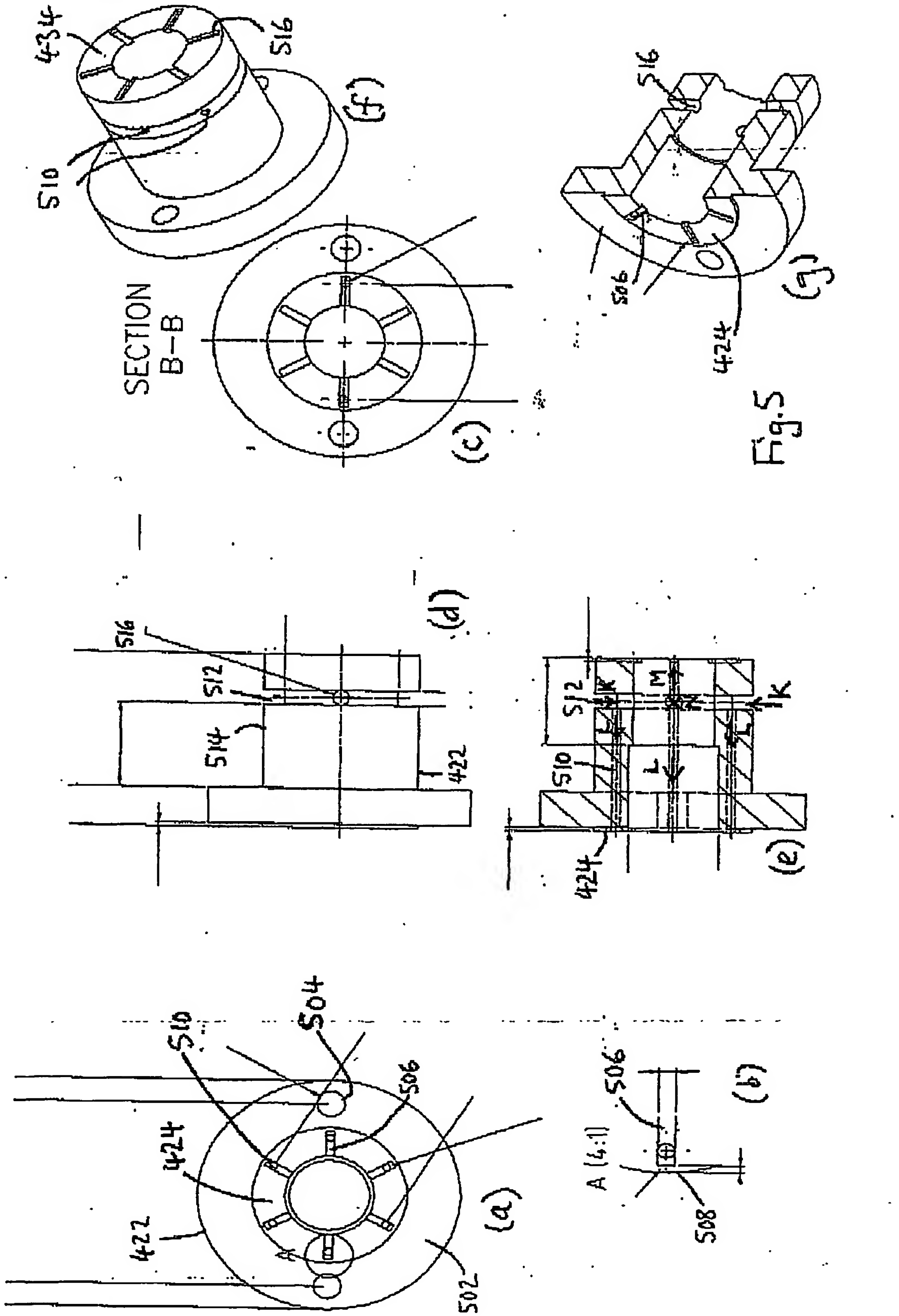
422

Fig. 4


434
516
510
(f)
SECTION B–B
(c)
516
506
424
(g)
Fig.5
516
512
514
422
(d)
512
M
K
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(e)
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422
502
(a)
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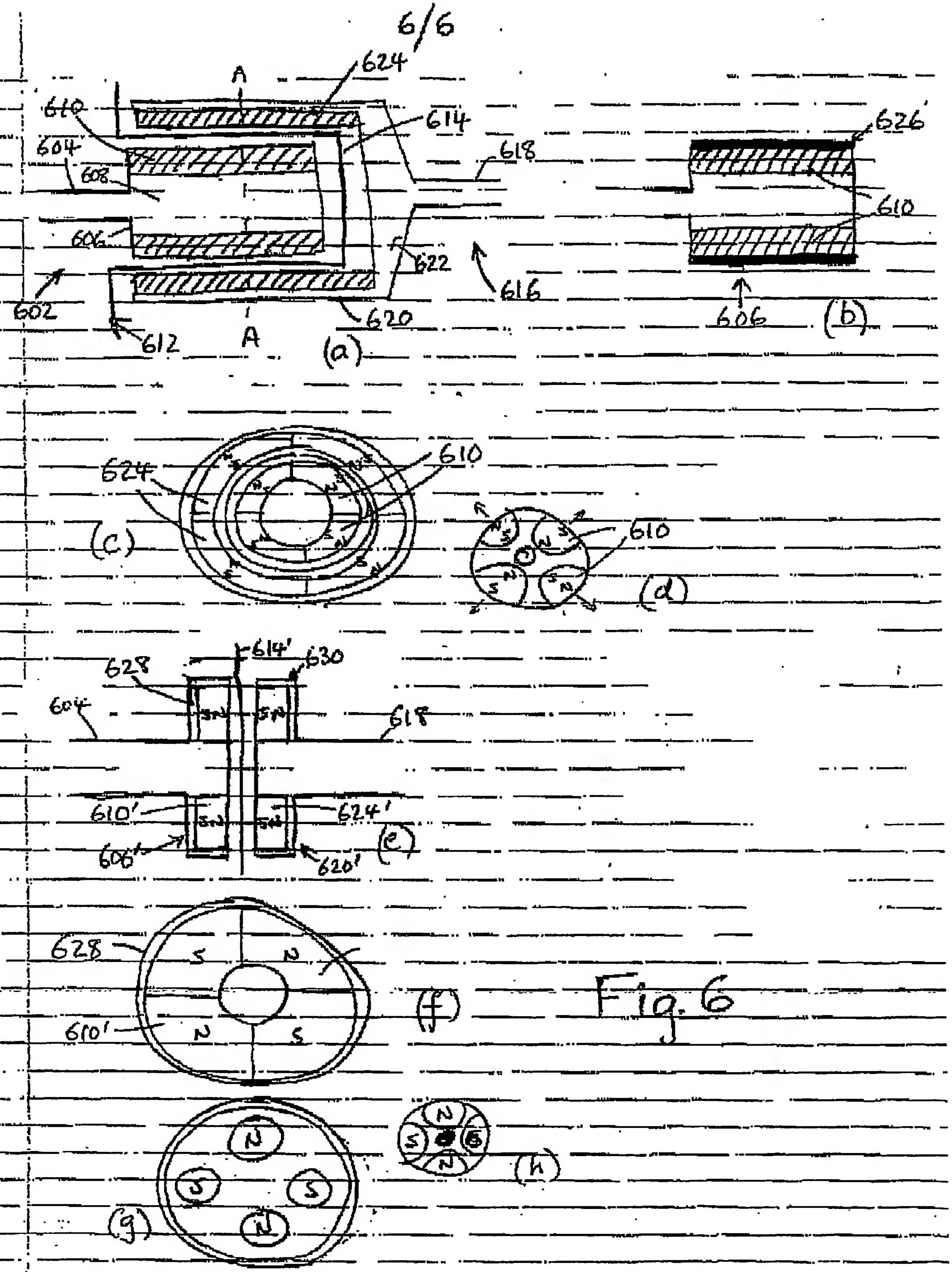
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624
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Fig. 6